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Theoretical analysis of a novel electricity–cooling cogeneration system (ECCS) based on cascade use of waste heat of marine engine

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ABSTRACT

This study presents a performance analysis of a novel electricity–cooling cogeneration system (ECCS) which combines a Rankine cycle (RC) with an absorption refrigeration cycle (ARC) to recover the exhaust heat of marine engine aboard ships. The RC is to provide electricity using water as the working fluid and the ARC is to provide cooling capacity needed aboard ships using a binary solution of ammonia–water as the working pair. Several performance parameters, including electricity output of Rankine cycle, cooling capacity, coefficient of performance (*COP*) of absorption refrigeration cycle and exergy efficiency, are analyzed and optimized. The simulation results indicate that compared with basic Rankine cycle, higher exergy efficiency can be realized by ECCS due to the extra cooling energy generated by the ARC, especially at a low condensing temperature of Rankine cycle. When the condensing temperature is 323 K and the superheat is 100 K, the exergy efficiency of the cogeneration system increases by 84%.

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1. Introduction

With increasingly world-wide concern on global warming and energy shortage, great effort has been carried out to improve the energy utilization and conversion [1]. Internal combustion engine (ICE) aboard ships is one of the major power output devices, whose effective thermal efficiency is only about 48–51% [2]. Therefore, capturing and reusing the waste heat of ICEs are of great significance to reduce energy consumption and emission and attract more attention [3].

Exhaust gas, scavenge air cooling, engine cooling jacket, and lubricating oil cooling (ranking in the decreasing magnitude) on ships are usually used as available waste heat sources to be recovered [4]. If the waste heat can be effectively recovered, fuel consumption and sailing cost will be greatly reduced. However, current proposals are mainly for space heating [5], heavy fuel oil heating [6] and ballast water heating [7], whose efficiencies are relatively low.

Current WHR technologies mainly include Stirling cycle, turbocharger/power turbine, distillation desalination, thermoelectric generators (TEG), Rankine cycle and absorption refrigeration cycle [8]. Due to its complicated mechanical arrangement [9] and transient response time [10], Stirling cycle engine has not been applied yet. Turbocharger/power turbine has become a well adopted WHR technology. Almost all medium and large diesel engines are equipped with a turbocharger since it increases the mass flow rate of air entering the engine without extra fuel consumption. Power turbine can implement continuous exhaust heat utilization for heavy duty engines, especially aboard large tonnage ships. ABB, Wsrtsila and MAN B&W companies have already equipped power turbine on some ships. The cost is low but a promising recovery ratio of 3–5% can be realized [3]. Distillation desalination is usually applied to recover the low temperature waste heat on ships, such as the jacket cooling water. Chen et al. [11] proposed that thermoelectric generator is promising to recover waste heat on ships and found that more than 600 W power may be produced from the waste heat of a 300 kW by means of simulation. However, its recovery ratio is relatively low.

Among all the WHR technologies, Rankine cycle is proposed to be the most promising technology, which has been widely applied in industrial WHR. Today, research on Rankine cycle to recovery waste heat of ICEs attracts more attention due to stringent emission regulations. Wang et al. [12] and Saidur et al. [13] presented reviews about waste heat recovery from ICEs by means of Rankine cycle. Different systems and working fluids were discussed in their papers. Ideas about the combination of ICEs and ORC were proposed in 1970s–1980s [14–17]. Literature survey indicated that presently Rankine cycle systems are still under development [18–19]. However, the temperature of ICE exhaust gas is much higher than the decomposition temperature of most organic fluids.

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Nomenclature

ARC	absorption refrigeration cycle
CMCR	contract maximum continuous rating
COP	coefficient of performance
ECCS	electricity–cooling cogeneration system
ICE	Internal combustion engine
MEPC	maritime environment protection committee
ORC	organic Rankine cycle
PER	primary energy ratio
RC	Rankine cycle
a, b	mass concentration of ammonia
c_p	specific heat
E_c	cooling exergy
E	exergy
h	enthalpy
m	mass flow rate
Q	heat energy
W	power
W_{net}	net power of RC
η	efficiency
ε	effectiveness of SHX

Subscripts

a	air
g	exhaust gas
G	electric generator
P	pump
T	turbine
w	water
0	environment
f	working fluid
ss	strong solution
ws	weak solution
r	refrigerant
c	cooling
in	input
out	output

In order to overcome this problem, water was usually used as the working fluid for recovery of medium or high temperature waste heat due to its high decomposition temperature, non-toxicity, environmentally friendly and low-cost, as indicated in previous studies by Parimal and Doyle [14] and Obieglo et al. [20]. Therefore, many ships have already mounted steam RC system to recover the waste heat of marine engines. The electricity generated by the steam RC system can be used for daily life or equipments of ship. However, in these systems the steam at the turbine outlet (usually over 100 °C) flows into condenser to be condensed directly, which led to a large amount of energy loss. Therefore, how to reuse the waste heat of steam in condenser is of great significance.

Cooling requirements aboard ships, such as air-conditioning, ice-making and medical or food preservation are usually driven by electricity, which increase the fuel consumption of ships. If cooling requirements can be satisfied by an ARC instead, fuel consumption will be greatly reduced because ARC can be powered by the waste heat of marine engines [3]. Currently lithium bromide–water and ammonia–water are conventionally adopted as working pair in ARC. The refrigerant of the former one is water and that of the latter one is ammonia. The advantage for refrigerant ammonia vapor (using ammonia–water) is that it can evaporate at lower temperatures (i.e. from −10 to 0 °C) compared to water (using bromide–water) (i.e. from 4 to 10 °C) [27]. Therefore, ammonia–water is selected as the working pair in ARC as it can satisfy the cooling requirement of lower temperature on ships [28]. Many investigations [21–26] have proposed that the heat source with temperature ranking from 50 °C to 200 °C is available to drive the ARC.

Therefore, the waste heat of steam at the turbine outlet is promising to be reused to power the absorption refrigeration cycle. Combination of Rankine cycle and absorption refrigeration cycle is of great research value because it can provide a greater useful energy output for a given fuel supply. Given its spacious room, stable operation condition and plenty of seawater as cold media, the cogeneration system is an excellent option to recovery the waste heat aboard ships.

In this paper, a novel ECCS based on RC-ARC is proposed. The RC is to provide electricity and ARC is to provide cooling. The potential of such a novel cogeneration system is explored. Calculations of both conventional steam RC only and the novel cogeneration system were conducted with EES (Engineering Equation Solver). The thermodynamic description of absorption cycle was based on the Schulz's equation of state for the ammonia–water [29] and the relative mathematical model was developed to descript the working process.

2. System descriptions

2.1. The topping marine engine

In the present study, the cogeneration system is driven by exhaust gas of 9RTA-96C Wartsila-Sulzer marine engine, whose rated power is 51,480 kW. The temperature and mass flow rate under tropical operation condition 100% CMCR (Contract Maximum Continuous Rating) load are 314 °C and 379,819 kg/h, respectively [30]. Based on the information of heavy fuel oil in Ref. [31], assuming the air–fuel ratio to be 18.5, the mass fraction of the exhaust gas is N_2 of 0.7216, CO_2 of 0.1760, H_2O of 0.0529 (gas), O_2 of 0.045 and SO_2 of 0.0045.

2.2. The bottoming cogeneration system

The proposed system consists of a steam Rankine cycle and an absorption refrigeration cycle, as shown in Fig. 1. The working principle of this combined cycle is described as the following.

Fig. 2 shows the T–S diagram of RC. The working fluid leaving the outlet of the condenser-1 (state 16) is pumped into the evaporator-1 (state17), in which the waste heat of exhaust gas heats the high-pressure liquid into superheated steam vapor (state 18). The

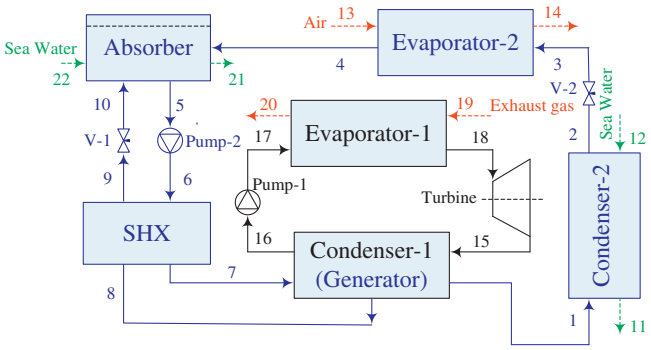

Fig. 1. Schematic of the novel ECCS.

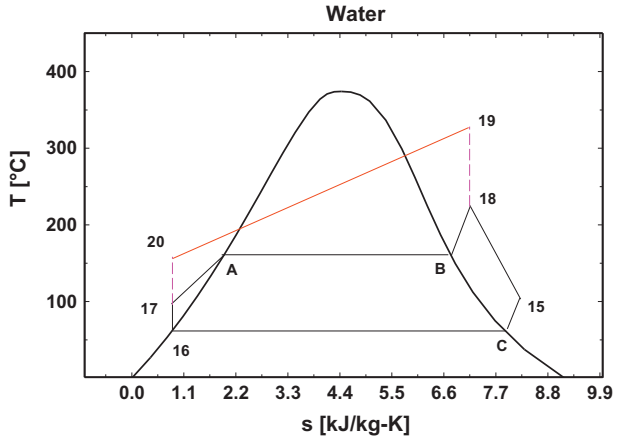

Fig. 2. T–S diagram of RC.

vapor then flows into the turbine to generate mechanical power. The power is then transferred to the electric generator or output shaft. At last, in the condenser-1, the steam at the turbine outlet (state15) transfers energy to the absorption refrigeration cycle and is condensed into liquid again, and a new cycle begins.

In the absorption refrigeration cycle, the waste heat of the steam at the turbine outlet in Rankine cycle heats the ammonia–water solution in the generator and separates ammonia vapor from solution. The rectified ammonia vapor then flows into condenser-2 and is condensed into liquid. Expansion valve V-2 is used to bring the condensing pressure down to evaporation pressure. Another function of V-2 is to control the mass flow rate of the refrigerant. Then the ammonia is completely evaporated into vapor in evaporator-2, generating cooling energy in this process. The vapor out of the evaporator-2 is then absorbed in absorber by weak solution that is depressurized by valve V-1. The strong solution from the absorber is pumped into the SHX by pump-2 and then into the generator. The cycle is finished.

3. Mathematics model

Computer simulations were conducted to evaluate the performance of the ECCS at different condensing temperatures of RC. For the purpose of the analysis, the following set of typical assumptions is considered:

(1) The system is operated at a steady state.
(2) The temperature of the exhaust gas out of evperator-1 is higher than the dew point.
(3) Before flowing into the condenser-2, the water content in the ammonia vapor is assumed to be removed thoroughly in a purification process.
(4) The weak solution at the outlet of the generator is saturated solution.
(5) The simulation program neglects pressure drops and heat losses.

Main operating parameters should also be set before the thermodynamic modeling as shown in Table 1.

A thermodynamic analysis must be performed in order to obtain each thermodynamic state in the ECCS. The analysis also involves the application of energy conservation and mass conservation.

3.1. Modeling of Rankine cycle

Based on assumptions and operating parameters mentioned above, the working process of each component can be described as follows:

Pump-1 (state: 16 → 17):

The fluid pump is driven by electricity and the electricity consumed by Pump-1 can be calculated:

$$W_{p1} = \dot{m}_f(h_{17} - h_{16})/\eta_{p1}$$

Evaporator-1 (state: 17 → 18):

This is an isobaric heating process in the evaporator-1. The heat transferred from the exhaust gas to the working fluid is equivalent:

$$Q_{eva1} = c_{pg}\dot{m}_g(T_{19} - T_{20})$$

$$\dot{m}_f(h_{18} - h_{17}) = c_{pg}\dot{m}_g(T_{19} - T_{20})$$

Turbine (state: 18 → 15):

Non-isentropic expansion process occurs in the turbine. The power generated by the turbine can be expressed as:

$$W_{out} = \dot{m}_f(h_{18} - h_{15})\eta_T$$

Condenser-1 (state: 15 → 16):

This is an isobaric condensation process in the condenser-1. The energy transferred in this process can be expressed as:

$$Q_{cond1} = \dot{m}_f(h_{15} - h_{16})$$

3.2. Modeling of refrigeration cycle

The working process of refrigeration cycle for each main component can be described as follows:

Generator (state: 7 → 1, 8) and (15 → 16)

The ammonia–water absorbed heat energy in generator. The energy balance and quality balance can be expressed as:

$$Q_{gen} = \dot{m}_f(h_{15} - h_{16})$$

$$\dot{m}_f(h_{15} - h_{16}) = \dot{m}_r h_1 + \dot{m}_{ws} h_8 - \dot{m}_{ss} h_7$$

$$\dot{m}_{ss} = \dot{m}_{ws} + \dot{m}_r$$

$$\dot{m}_{ss} a = \dot{m}_{ws} b + \dot{m}_r c$$

Condenser-2 (state: 1 → 2):

In the condenser-2, the seawater can be taken as a constant cold media. The energy balance and quality balance can be expressed as:

Table 1
Main parameters of operating conditions.

Parameters	Value	Parameters	Value
Atmosphere pressure	101 kPa	Mole fraction of strong solution out of absorber	0.7
Atmosphere temperature	303 K	Min temperature difference in the condenser of ARC	6 K
Sea water temperature	288 K	Min temperature difference in the evaporator of ARC	5 K
Efficiency of pump	0.8	Min temperature difference in the SHX	10 K
Efficiency of generator	0.8	Min temperature difference in the absorber	6 K
Efficiency of turbine	0.7	Condensing temperature of refrigeration cycle	294 K
Outlet temperature of exhaust gas	393 K	Evaporation temperature of refrigeration cycle	273 K

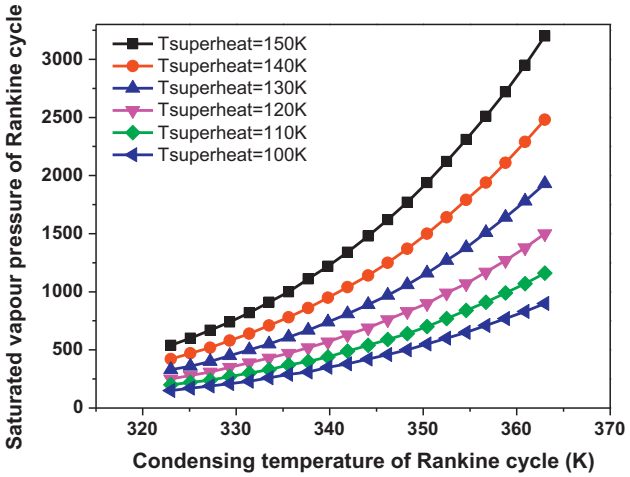

Fig. 3. Influence of condensing temperature on the saturated pressure.

$$\dot{Q}_{cond2} = m_r(h_1 - h_2)$$

The function of expansion valve V-2 is to control both the evaporation pressure and the mass flow rate of refrigerant. It can be taken as an adiabatic process.

$$h_2 = h_3$$

Evaporator-2 (state: 3 → 4) and (13 → 14):

The evaporator-2 is a component that provides cooling energy.

$$Q_{eva2} = \dot{m}_r(h_4 - h_3)$$

$$\dot{m}_r(h_4 - h_3) = c_{pa}\dot{m}_{air}(T_{13} - T_{14})$$

Absorber (state: 4, 10 → 5):

The ammonia vapor flows into absorber and is absorbed by the weak solution.

$$Q_{abs} = \dot{m}_r h_4 + \dot{m}_{ws} h_{10} - \dot{m}_{ss} h_5$$

$$\dot{m}_r + \dot{m}_{ws} = \dot{m}_{ss}$$

$$\dot{m}_r c + \dot{m}_{ws} b = \dot{m}_{ss} a$$

Pump-2 (state: 5 → 6):

The fluid pump is driven by electricity. The electricity consumed by pump-2 can be expressed as:

$$W_{p2} = \dot{m}_{ss}(h_6 - h_5)/\eta_{p2}$$

SHX (state: 6 → 7) and (8 → 9):

Solution heat exchanger (SHX) is also named economizer, which can improve the thermal efficiency of refrigeration cycle. The energy conservation can be expressed as:

$$\dot{m}_{ss}(h_7 - h_6) = \dot{m}_{ws}(h_8 - h_9)$$

As the solution valve is adiabatic, the following equation can be expressed:

$$h_9 = h_{10}$$

3.3. Evaluation of the system performance

Thermal efficiency η and coefficient of performance (*COP*) are adopted as important parameters to evaluate the performance of Rankine cycle and refrigeration cycle, respectively. η is defined as electricity output of turbine divided by energy input, given by

$$\eta = W_{out}/(Q_{eva1} + W_{p1})$$

COP is defined as cooling capacity output divided by energy input. In order to calculate the net *COP*, power consumption by the pump was taken into account.

$$COP = Q_{eva2}/(Q_{gen} + W_{p2}/\eta_{p2})$$

Primary energy rate [32] (PER) is usually used to evaluate the performance of cogeneration system. However, there is a great difference between the energy grade of electricity and cooling energy. While considering the difference of energy grades, exergy efficiency is usually used to evaluate the performance of cogeneration system. According to Ref [33], exergy efficiency is generally defined as the exergy output divided by the exergy input:

$$\eta_{ex} = \frac{W_{net} + E_c}{E_{in}}$$

$$W_{net} = W_{out} - W_{p1}$$

$$E_c = \left(\frac{T_0}{T} - 1\right) Q_{eva2}$$

$$E_{in} = E_{19} - E_{20} + W_{p1} + W_{p2}$$

$$E_{19} = ((h_{19} - h_0) - T_{13}(s_{19} - s_0))m_g$$

$$E_{20} = ((h_{20} - h_0) - T_{13}(s_{20} - s_0))m_g$$

Wherein, the η_{ex} is the exergy efficiency, E_{in} is the total exergy input in the cogeneration system, W_{net} is the net power output in RC and E_c is the exergy associated with the cooling energy.

4. Results and discussion

4.1. Performance of basic Rankine cycle

As water is a wet working fluid, it is important to make sure that the steam at the turbine outlet is dry to prevent the turbine from corrosion. However, the temperature of heat source is fixed to be 314 °C only. Corrosion problem during the expansion process can be prevented by controlling two significant parameters, including the mass flow rate and the evaporation pressure. In the presented work, the mass flow rate is determined by means of pinch point method [34]. Main details are described by Vaja and Gambarotta [35]. In order to explore the potential of this cogeneration system, the maximum cycle pressures under different condensing temperatures are calculated. For a given superheat temperature and condensing temperature, there is a saturated vapor pressure. The cycle pressure must be lower than the saturated vapor pressure provided by Fig. 3. Otherwise, the steam at the turbine outlet will partially become liquid, which will cause corrosion on turbine blades.

Fig. 3 presents the effect of condensing temperature on the saturated vapor pressure (also named vapor pressure), which is the pressure of a vapor in equilibrium with its non-vapor phases. It is apparent that the saturated vapor pressure increases with the increase of condensing temperature and superheat temperature. According to the T–S diagram of water, with the increase of the condensing temperature, the state of saturated vapor curve (point C) shifts to the left continuously, which allows a higher pressure for the saturated vapor. As the superheat of steam increases, the steam temperature at turbine inlet (T18) increases. It means that for a given condensing temperature, the state point at the turbine outlet (point 15) goes far away from the saturated vapor curve, which allows a higher saturated vapor pressure. Generally speaking, the Rankine cycle must be operated at a pressure lower than the saturated pressure provided by Fig. 3.

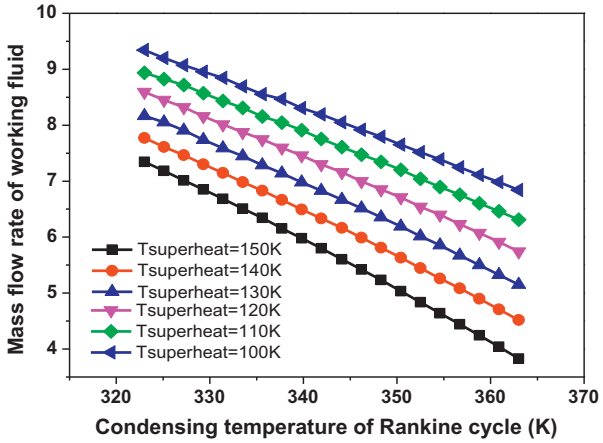

Fig. 4. Influence of condensing temperature on mass flow rate of working fluid.

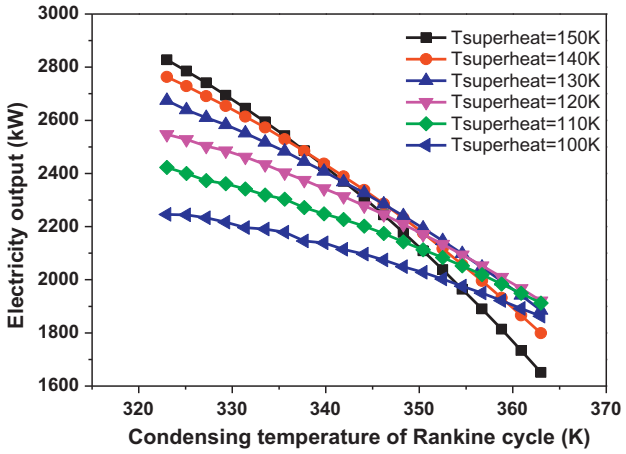

Fig. 5. Influence of condensing temperature on electricity output.

There is a maximum mass flow rate of working fluid for each condensing temperature. As shown in Fig. 4, the mass flow rate of Rankine cycle decreases evidently and linearly with the increase of condensing temperature. In other words, with the increase of mass flow rate, the vaporization temperature and pressure decreases. The condensing temperature needs to be lower to avoid the corrosion of turbine. In the Rankine cycle, the superheat is another parameter that influences the mass flow rate. It is further observed in Fig. 4 that the mass flow rate of working fluid decreases with the increase of superheat. In other words, the mass flow rate should be lower to realize a higher superheat.

Fig. 5 shows that the electricity generated by the electric generator of Rankine cycle decreases with the increase of condensing temperature. This is mainly due to the fact that the energy contained in the steam at the turbine outlet is larger with a higher condensing temperature, which will lead to less energy conversion in the turbine. It is better to set a lower condensing temperature to produce more electricity. Taking pressure into consideration, a higher condensing temperature corresponds to a higher condensing pressure. Therefore, the pressure difference between the turbine inlet and outlet is lower. The conversed energy is also lower compared to that with a lower condensing temperature. It is further observed that with a higher superheat, the decline rate of the electricity is much larger than that with a lower superheat. From the modeling we know that the generated electricity depends on both the mass flow rate and the enthalpy difference of steam between turbine inlet and turbine outlet. As mentioned above, the mass flow rate is lower with a higher superheat. With a higher superheat, the enthalpy difference varies significantly as the condensing temperature varies. This is the main reason why the decline rate of the electricity at superheat = 150 K is the highest in this study.

As shown in Fig. 6, the thermal efficiency increases with the increase of condensing temperature. The thermal efficiency of the Rankine cycle is defined as electricity divided by energy input in RC. As RC is a closed loop, the condensing temperature of the working fluid is closely relative to its evaporation temperature. With a higher condensing temperature, the temperature of working fluid at evaporator-1 inlet increases. The temperature difference between the working fluid and the exhaust become smaller [36], which prevents too much energy from being wasted in the condenser-1. Furthermore, the working fluid absorbs less energy during evaporation process. Therefore, although the electricity output decreases with the increase of condensing temperature as mentioned above, the thermal efficiency increases.

4.2. Performance of refrigeration cycle

In the cogeneration system, the mass flow rate and temperature of ammonia–water solution are variables to adjust the condensing temperature of RC. As shown in Fig. 7, the mass flow rate of ammonia vapor from the generator decreases with the increase of condensing temperature of RC. With a higher condensing temperature of RC (corresponding to the generating temperature of ARC), the ammonia vapor separated from per unit mass of strong solution is more than that with a lower generating temperature. However, the mass flow rate of strong solution flowing into the generator was adjusted to a lower level. Conversely, although the ammonia vapor separated from per unit mass of strong solution is less with a lower generating temperature, the mass flow rate of strong solution increases. Fig. 7 shows that the ammonia vapor quantity separated from the evaporator-2 is more with a lower condensing temperature than that with a higher condensing temperature of RC. Therefore, to select an appropriate condensing temperature is important for both electricity output of RC and cooling capacity of ARC.

Fig. 8 shows that the cooling capacity of the ARC decreases with the increase of condensing temperature of RC. It has to be mentioned that when the condensing temperature of RC increases, the ammonia vapor separated in the generator declines. Because the condensing temperature, condensing pressure and refrigeration temperature of refrigeration cycle are all constant, the cooling capacity is determined by the mass flow rate of refrigerant

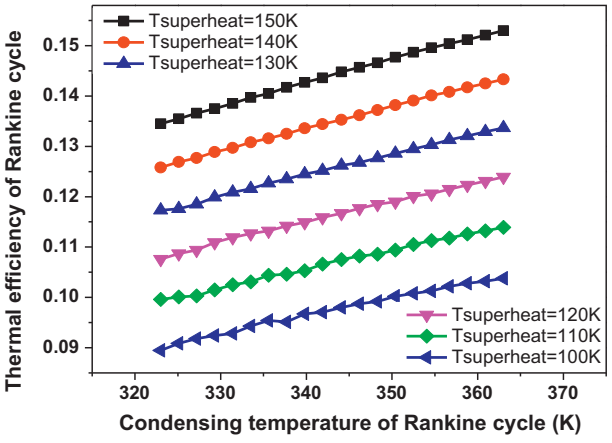

Fig. 6. Influence of condensing temperature on thermal efficiency.

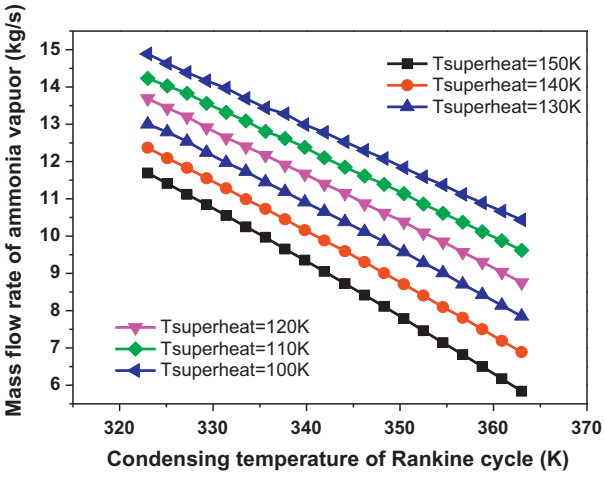

Fig. 7. Influence of condensing temperature on mass flow rate of ammonia vapor.

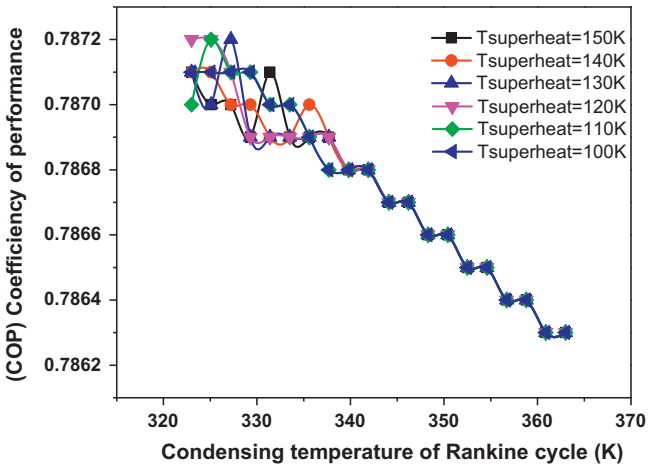

Fig. 9. Influence of condensing temperature on *COP*.

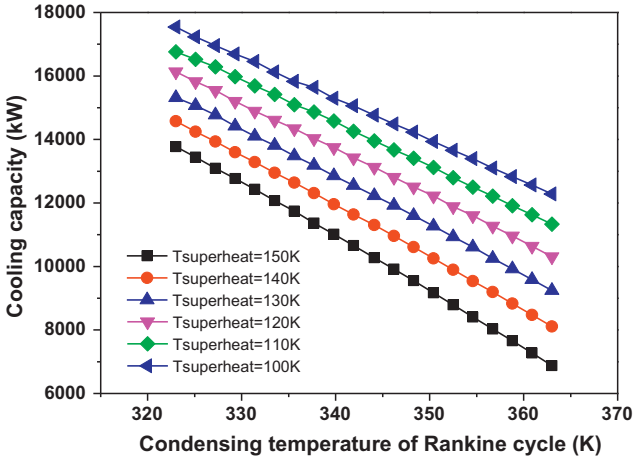

Fig. 8. Influence of condensing temperature of RC on the cooling capacity.

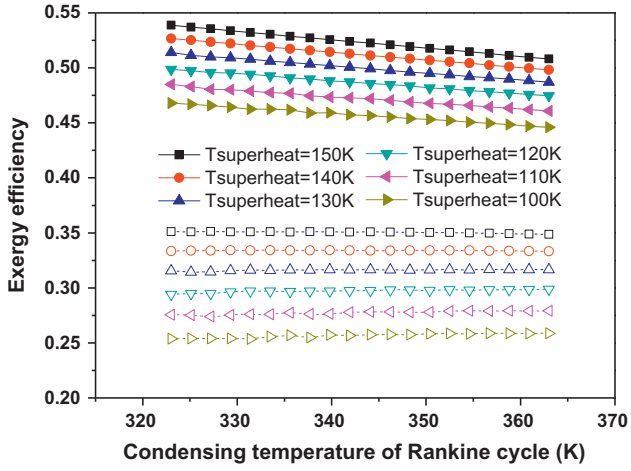

Fig. 10. Comparisons of exergy efficiency between basic RC and cogeneration system.

(ammonia vapor). That is to say, the variation trend of the cooling capacity of refrigeration cycle is the same with that of mass flow rate of ammonia vapor.

The *COP* is defined as the cooling effect capacity divided by the energy input in refrigeration cycle. Fig. 9 shows that the difference of *COP* between different condensing temperatures of RC is minor, less than 0.001. The *COP* of the absorption refrigeration cycle has a fluctuating downtrend as the condensing temperature of RC increases. As mentioned above, the cooling capacity becomes less as the condensing temperature of RC increases. In addition, the energy transferred in the generator is less due to a smaller mass flow rate of ammonia–water solution and a lower temperature difference between the steam and ammonia–water solution. The effects of these two parameters lead to the fluctuation of *COP* when the condensing temperature is in the range of 325–335 K.

4.3. Performance of cogeneration cycle

Exergy efficiency is a key parameter to evaluate the performance of the cogeneration system. The difference between different energy grades is taken into consideration with this method. Results in Fig. 10 clearly show the benefits of the cogeneration system compared to basic Rankine cycle only. In this cogeneration system, both electricity and cooling can be provided by recovering waste heat of marine engine. The total output determines the performance. As shown in Fig. 10, the effect of superheat temperature of RC is larger than that of condensing temperature. Compared to basic RC only, the exergy efficiency of the cogeneration system improves significantly. The improvement is more evident at a lower condensing temperature of RC. When the condensing temperature is 323 K and superheat temperature is 100 K, the exergy efficiency increases from 0.2538 to 0.4679. As a result, the combination of RC and ARC can lead to significant improvement, lowering the fuel consumption of the marine engine.

5. Conclusions

In this investigation, theoretical analysis of a novel ECCS driven by exhaust gas of a marine engine is presented. Model calculations were performed at different condensing pressures and superheat temperatures of RC. The electricity output increases with the increase of superheat of working fluid, and decreases with the increase of condensation temperature. Meanwhile, the cooling capacity decreases with the increase of condensing temperature and superheat of working fluid. Higher exergy efficiency can be achieved while recovering the energy of expanded steam at turbine outlet to generate cooling energy. When the condensing temperature is 323 K and superheat temperature is 100 K, the exergy

efficiency increases by 84%. It is obvious that both the electricity output and cooling capacity can be improved at a lower condensing temperature. Moreover, the cogeneration system has a better performance than the basic steam RC only.

In total, it is recognized that this ECCS possessed great potential to recover the waste heat energy of marine engine. In order to realize the application of this novel cogeneration system, further investigation will be carried out in future work.

Acknowledgments

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